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EXPERIMENTAL INVESTIGATION OF THE VERTICAL FORCES ACTING ON PROLATE SPHEROIDS IN SINUSOIDAL HEAVE MOTION

by

Thomas C. Watson

Under

Contract Number Nonr-222(30)

SERIES NO. 82

ISSUE NO. 18

DATE April 1961

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ABSTRACT

A planar motion generator for constraining a small ship model to pure sinusoidal motion in any one of several degrees of freedom plus combinations of certain of these, was designed and constructed for use in the Ship Model Towing Tank of the University of California. The apparatus, briefly described here, was tested for operation in heave motion only. Presented herein are the results of vertical force measurements made for each of three horizontal, prolate spheroids oscillating at various frequencies in a free surface and at different depths of submergence. The measured forces for the surface tests are compared with results obtained by strip calculations. Theoretical values of added mass and damping for a horizontal, semi-immersed, circular cylinder of infinite length oscillating in heave were used in the strip calculations.

INTRODUCTION

The need for research apparatus designed specifically for the experimental study of hydrodynamic forces and moments which act on ship models undergoing various types of motion has been clearly expounded by authorities such as Weinblum [1], Golovato [2], and Vossers [3]. One such piece of equipment which was developed as a result of this need is the planar-motion mechanism used in forced heaving and pitching studies of large ship and submarine models at the David Taylor Model Basin. Construction and operation features of the machine are described in great detail by Gertler [4]. Weinblum, Brooks, and Golovato [5] describe a different type of mechanism used in experimental investigations of free-heave oscillations of ship models and submerged bodies. While the free-heave oscillators are more simply constructed, the forced-motion mechanisms are more versatile research tools because, to mention one advantageous feature, they afford a ready means for investigation over a wide range of frequencies. Also the hydrodynamic problem associated with forced oscillation is more easily and accurately solved, thus fostering an unambiguous interpretation of experimental results. The free heaving motion is only approximately periodic, contrary to the usual assumption of periodicity made in the theoretical analysis of the problem. Because of the preceding and other favorable arguments, the forced-motion technique is usually favored in present-day experimental investigations of forces due to ship-model oscillations.

A primary objective in the experimental investigation of the forces and moments which act on bodies oscillating in a fluid is to obtain information from which the added mass and damping coefficients can be extracted. These coefficients are essential for a more precise expression of the equations of motion for a body which experiences unsteady motion in a fluid. Until recently theoretical values for added mass and damping coefficients of even simple two-dimensional bodies undergoing heave oscillations in a free surface were known only for the limiting cases where the frequency approached either infinity or zero. Ursell [6] determined theoretically the hydrodynamic forces acting on a semi-immersed horizontal circular cylinder of infinite length in heave motion and thereby obtained the desired coefficients. More recently Porter [7] developed a theoretically exact procedure whereby the frequency-dependent added mass and damping coefficients in heave can be calculated for horizontal, floating cylinders of arbitrary cross section and infinite length, which are symmetric about a longitudinal vertical plane. Havelock's solution [8] for a sphere heaving in a free surface is the only three-dimensional case for which the coefficients are known exactly. Most attempts to derive the coefficients for a three-dimensional body are based upon the assumption that the added mass and damping coefficients for two-dimensional sections can be applied to segments of the body in a strip calculation. Some sort of correction factor then is applied to correct for the three-dimensionality of the flow.

Vossers [3] has presented an up-to-date summary of the theoretical and experimental investigations of added mass and damping for ship forms and other bodies. Golovato [2], Weinblum [5], and Porter [7] are among those who have provided experimental data that is readily compared with theoretical results for bodies undergoing heave and/or pitch oscillations in a free surface.

The primary purpose of this paper is to present the results of an experimental investigation on the frequency dependence of the vertical forces acting on each of three horizontal, prolate spheroids constrained to small sinusoidal heave oscillations in a free surface and at various depths of submergence. A secondary purpose is to briefly describe the sinusoidal planar-motion generator used in the experimental force investigation. The agreement observed between measured and calculated heaving forces on a semi-immersed circular cylinder [7] and on three submerged spheroids may be regarded as evidence that the motion generator performs its intended function in a reasonably successful manner.

EXPERIMENTAL APPARATUS

The experimental equipment consisted of a planar-motion generator, two single-component force balances, three prolate spheroids, and the requisite electrical recording apparatus. All tests were conducted in the Ship Model Towing Tank (University of California) which is 200 feet long, 8 feet wide, and can be filled to a depth of 6 feet.

The motion generator is essentially two Scotch-yoke mechanisms driven by a continuously variable-speed transmission and constant-speed a-c motor. The frequency was variable from 0 to 3.6 cps for some of the tests described herein, but this range can be varied by changing the fixed gear ratio between the transmission and the drive shaft of the Scotch yokes. The function of each Scotch yoke is to convert the constant-speed rotary motion of a shaft into sinusoidal axial motion of a rod. By attaching the motion generator to a horizontal model at two points, it is possible to obtain either pure heave, pure pitch, or a combination of heave and pitch motions. The resulting motion of the model depends upon the phasing between the motions of the two reciprocating rods; i.e., pure heave is obtained when the phase is zero or pure pitch when the phase is 180 degrees. The elevation of each rod is individually controlled to permit leveling of the attached model. The amplitude of the motion of each rod is continuously variable from 0 to 1.5 inches. For heaving and/or pitching tests at zero forward speed the motion generator is mounted on the towing carriage with the axes of the reciprocating rods vertical and with the longitudinal axis of the model normal to the tank walls. Figure 1 is a photograph of the motion generator with a spheroid installed in the heave and pitch test position.

By suitable orientation of the motion generator and use of special attachment brackets, it is possible to obtain the other four modes of ship motion; i.e., sway, yaw, roll, and surge. Detailed descriptions of the design, construction, and operation of the motion generator are presented elsewhere [9,10].

Two identical force balances used for the series of measurements described herein were designed to respond to only one component of force. The balances are essentially flexural beam assemblies, each consisting of a pair of rigid parallel plates joined by four cantilever beams. Each assembly is milled from a stainless steel cube 1.5 inches on each edge. For some of the force measurements made on the spheroids heaving in the free surface, a linear differential-voltage transformer was used to detect the relative in-plane motion of the two plates of the force balance. For the remaining tests small strain gages (Baldwin SR-4, Type A-8) were cemented to the cantilevers and made waterproof. In heave tests the plates of each force balance are vertical, with the model attached to one plate and the towing strut and reciprocating rod attached to the other. Static calibrations demonstrated that the force balances possessed linear operating characteristics over a ± 10 pound range. The spring constant of each force balance was found by measurement to be 7500 pounds per inch. Because the spring constant was high, considering the maximum heave forces measured, the deflection of the beam system was negligible in comparison with the amplitude of the constrained heave motion. Bench tests demonstrated that the force balances were highly insensitive to horizontal forces acting normal to the beams, but that they were sensitive to horizontal forces aligned with the beams. Consequently, the balances were mounted in the models with the beams directed athwartship.

Prolate spheroids made of wood were selected as the test bodies in the experimental study for a number of reasons. Not the least important from a practical viewpoint is the fact that such bodies of revolution can be readily and accurately produced on an ordinary machine shop lathe. In addition, the added mass of a spheroid oscillating in an infinite fluid is well known [11] from ideal fluid flow theory. Utilizing this knowledge, a comparison between theoretical and experimental results could be made for the case of the deeply submerged spheroids. Furthermore, vertical forces acting on heaving semi-immersed spheroids are exceptionally well suited to a strip-type calculation, since both the added mass and damping coefficients are theoretically known [6,7] for a semi-immersed circular cylinder heaving at various frequencies.

The volume of each of the three spheroids was the same, but the length-beam ratio differed in each case, being 4, 6, and 8. Important geometrical quantities of the three models and the experimental setup are shown in Figure 2. The models were heavily coated with a resin-based waterproofing to prevent changes of mass due to water absorption during the test period.

Each model had recessed wells, symmetrically spaced fore and aft, in which the force balances and lead ballast were installed. Since the force balances were between the model and the towing struts, horizontal forces exerted on the struts had no influence on the force measurement. Each towing strut passed through a fairing block, which was mounted

in position before the model was turned on a lathe. Thus a continuous surface of revolution was maintained for the submerged tests, except for a small opening where the towing struts entered the fairing blocks. Three of the wells and one fairing block are clearly shown in Figure 3, which is a photograph of the spheroids. The towing struts were bodies of lenticular section constructed by welding together two strips cut longitudinally from a stainless steel circular tube.

The electronic equipment consisted of a number of amplifiers for monitoring the several electrical signals of interest. These signals came from the two force transducers and a position-indicating potentiometer. An eight-channel pen oscillograph utilizing heat-sensitive chart paper was used to record the electrical signals. Each amplifier channel was equipped with a carrier oscillator and demodulator for use with the strain gages of the force balances. The position-indicating potentiometer was rigidly attached to one of the reciprocating rods and driven by a taut steel wire wrapped around the shaft and fixed relative to the base of the motion generator. Comparison of the signal from the potentiometer and a low-frequency voltage oscillator revealed that the motion of the reciprocating rod was indeed sinusoidal.

EXPERIMENTAL PROCEDURE

The vertical forces caused by small heaving oscillations of the spheroids were measured under three sets of test conditions: the models were semi-immersed with zero forward speed, semi-immersed with different forward speeds, and submerged with zero forward speed. For all tests the models were ballasted to approximately neutral buoyancy, since the force balances were not designed to operate under a large static force. The weight of each model was determined with a balance scale whose resolution was 0.01 pound.

The force balances were calibrated and checked for linear response after installation by the addition of a series of accurately known weights to the models. Appropriate corrections were made for the weights applied under water to calibrate the force balances for the submerged runs.

It has already been pointed out that the longitudinal axis of the model was oriented normal to the tank walls for the semi-immersed tests at zero forward speed. No attempt was made to assess the influence of waves reflected from the tank walls. Each force measurement, corresponding to oscillation of the model at a preset frequency, was of sufficiently short duration that it was completed before waves reflecting from the ends of the tank could reach the model. For each test the model was oscillated only until visual observation of the recorded force signals indicated that steady-state conditions definitely had been established. All the submerged runs at

zero forward speed were also made with the axes of the models normal to the tank walls. The semi-immersed towed tests were made with the longitudinal axis of the model parallel to the direction of travel. The primary object of the towed runs was to demonstrate that the motion generator and electrical equipment functioned properly while the towing carriage was in motion, rather than to make a systematic investigation of the forces acting on a heaving spheroid with forward motion. The amplitude of oscillation was adjusted for accuracy with a machinist's dial indicator graduated in thousandths of an inch. Dial indicators also were used as an accurate means to determine when the phase between the reciprocating rods was adjusted to zero. This was necessary to ensure that pure heave motion occurred.

For all tests the water depth was 4.5 feet. Some of the initial tests with the spheroid designated B (L/B = 6) in Figure 2 were made between a pair of false walls installed parallel to the sides of the tank at midlength. These false walls were 8 feet long, spaced 42 inches apart, and rose vertically from the tank bottom to above the free surface. Additional walls 81 inches long reached diagonally from the ends of the false walls to the sides of the tank. For some oscillation frequencies, corresponding to a band near the natural frequency of transverse oscillation of the water in the tank, wave interference was noticeable in the force measurements. For other frequencies of oscillation the waves appeared to radiate primarily toward the tank ends with little energy stored or radiated in waves across the tank.

RESULTS OF FORCE MEASUREMENTS

The vertical force data obtained from the measurements made on all surface tests are plotted with normalized force as ordinate and frequency of oscillation as abscissa. Normalized force is defined as the ratio of the measured force amplitude (F_o) to the change in hydrostatic buoyancy force (ρgAa_o). The data for the submerged tests are plotted with the force amplitude as ordinate and the frequency as abscissa. In all the figures contained herein V denotes model towing speed in knots; a_o is the amplitude of heaving motion in inches; L/B is the length-beam ratio of the spheroid; H is the depth of submergence of the spheroid's longitudinal axis beneath the still water surface.

As indicated earlier, the initial tests were made with the axis of Model B oriented perpendicular to a pair of false walls. As the model was oscillated in heave at various frequencies and zero forward speed in the free surface, the vertical forces caused by the motion were measured and recorded. Since no discernible differences were noted when the set of tests was repeated with the false walls removed, a distinction is not made in the data presented in Figure 4 for the two different sets of tests. There was less scatter of the data when sufficient time elapsed between runs for the wave action to become neglibible. Only three to five minutes were required because an efficient wave absorber is installed in one end of the tank. The surface tests with Model B were repeated for

three different amplitudes of oscillation: 1/8, 1/4, and 1/2 inch. Frequency of oscillation ranged from 0 to about 2.5 cps. The lack of consistent significant differences in behavior of the force measurements for the three amplitudes of oscillation could indicate that the variation of oscillation amplitudes did not extend outside the range of motion within which the linearized theory is valid, even though the ratio of the largest amplitude (0.5 inch) to the maximum diameter of the spheroids (7.55 inches) does not appear to be particularly small.

The other two spheroids, Models A and C, were subjected to similar surface tests, but without the false walls and with a single heaving amplitude of 0.25 inch. Results of these tests for the two models are shown in Figures 5 and 6.

Only one amplitude of oscillation (0.25 inch) was used for the submerged force measurements on Model B. The results of these tests are presented in Figure 7 where the total vertical force (sum of the forces measured by the two force balances) is plotted for different frequencies of oscillation. Depths of submergence were equal to 1/2, 1, 3/2, and 3 maximum beam widths of the spheroid. Similar vertical force measurements were made for the other two spheroids when submerged at various depths. An amplitude of oscillation equal to 0.25 inch was also imposed for these tests. Data for the submerged runs on Models A and C are presented in Figures 8 and 9. The data marked by + in Figure 9 is for a depth of submergence slightly greater than one-half the maximum beam of Model C.

Theoretical curves, with vertical force plotted as a function of frequency, are also presented for the submerged tests on each model. These curves were calculated under the assumption that no damping occurred and that the values of added mass for spheroids oscillating in an infinite fluid were applicable. Under these assumptions the theoretical curves are parabolas. It can be observed that the measured and theoretical forces do not differ greatly throughout the range of investigation.

The trend of results obtained for oscillation in the free surface is about as well defined as can be expected, judging from the comments made in references [3] and [5] about such tests at zero and very slow forward speeds. There it is pointed out that wave reflections seriously impair the validity of force measurements made on towed models in narrow tanks where the ratio of model speed to wave speed is less than 1/4.

No attempt was made to quantitatively compare the force measurements made here with those made by other investigators. It can be readily observed that the general shape of the plotted data is similar to that obtained by Golovato for force measurements made on a heaving ship model in a large basin where wave reflections were less troublesome. Golovato's data also exhibit no pronounced dependence upon the amplitude of oscillation. The ratio of maximum amplitude of oscillation to maximum beam width of the models was approximately the same for Golovato's and the present experimental investigations.

Weinblum [5] presents the results of an experimental investigation of the added mass and damping in heave for a prolate spheroid of length-to-beam ratio equal to 7. Since none of the force measurements reported herein were resolved into in-phase and quadrature components, the data cannot be compared except for the deeply submerged tests at zero forward speed. Weinblum reports that when the spheroid was oscillated at a depth equal to 2.9 beams, the added mass coefficient was very near the value given by Lamb, namely 0.93, for the case of oscillation in an infinite fluid. A similar conclusion was reached when the three spheroids were oscillated in this experimental investigation at depths equal to 3.0 diameters. The close agreement of the experimental and theoretical values displayed in Figures 7, 8, and 9 is taken as evidence of the validity of Lamb's values [11] for the added mass.

The results of measuring the vertical heave force when Model A was towed semi-immersed at three different speeds are presented in Figure 10. Although the data are more widely scattered than for the tests made at zero forward speed, the major trend is distinct and not greatly different from that observed when the forward speed was zero. Heaving amplitudes were 1/4 inch for all towed tests. According to Vossers [3], among others, when the ratio of towing speed to wave speed is less than 1/4, a progressive wave system extends ahead of the model, as well as athwartship and aft. A portion of the wave pattern which progresses in the forward direction reflects back into the path of the oncoming model. The wide scatter and

large variations of the force data for the lower frequencies can be explained in view of this wave reflection phenomenon. Less scatter of the data is noted for the tests made at higher frequencies. In this higher frequency range the ratios of towing speed to wave speed were greater than 1/4, indicating that the wave system does not extend ahead of the towed model.

STRIP CALCULATIONS

Contained in Figures 4, 5, and 6 are calculated curves of the normalized forces acting on each of the three semi-immersed spheroids. These three curves, which are reproduced in Figure 11 for the sake of comparison, result from strip calculations. The calculations were adapted from the method given by Porter [7] for determining the normalized force per unit length of an infinite circular cylinder heaving in a free surface. It is of some interest to point out the several assumptions upon which the analysis rests and to show how the strip calculations proceeded: (1) damping due to viscosity of the fluid is negligible in comparison with that due to radiation of the wave energy; (2) the cylinder is constrained to small, harmonic heaving oscillations; (3) the only other forces which act on the spheroid are due to hydrostatic buoyancy, the hydrodynamic or added mass effect, and the inertia of the mass of the heaving body. Under these assumptions the linear differential equation for vertical heaving motion is

$$\ddot{y} + C\dot{y} + Dy = E \sin (\omega t + \phi) \qquad (1)$$

where

$$C = \frac{2h\omega}{K + k}$$

$$D = \frac{\rho g A}{m(K + k)}$$

$$E = \frac{F_o}{m(K + k)}$$

In the above equations the symbols have the following meanings:

y	Vertical coordinate with origin in still water surface
$2h$	damping coefficient
$k=m'/m$	added mass coefficient
$K=M/m$	body mass coefficient
m'	added mass
M	mass of heaving body
ρ	fluid density
g	acceleration due to gravity
A	water plane area of the heaving body
ω	angular frequency
ϕ	phase angle of force with respect to motion
t	time in seconds
F_o	amplitude of vertical heaving force

We wish to determine F_o in terms of the other parameters appearing in equation (1), since the force amplitude is the quantity measured in the experimental investigations. The displacement is constrained by the motion generator to be of the form

$$y = a_o \sin \omega t \qquad (2)$$

where a_o is the amplitude of oscillation. Substitution for y

and its derivatives in equation (1) yields

$$G \sin \omega t + H \cos \omega t = E \sin (\omega t + \phi) \tag{3}$$

where

$$G = a_o(D - \omega^2)$$

$$H = a_o \omega C$$

Introducing a new constant defined by

$$\tan \phi = \frac{\omega C}{D - \omega^2} \tag{4}$$

We can rewrite equation (3) as

$$a_o \left[(D - \omega^2)^2 + (\omega C)^2\right]^{1/2} \left[\sin \phi \cos \omega t + \cos \phi \sin \omega t\right] = E \sin(\omega t + \phi)$$

Making use of a trigonometric identity and reverting to the original parameters, we obtain from the above equation, upon dividing through by the change in hydrostatic buoyancy force per unit length, the normalized force per unit length of the heaving cylinder

$$\frac{F_o}{\rho g a_o A} = \left\{\left[1 - \frac{m\omega^2(K + k)}{\rho g A}\right]^2 + \left[\frac{2h\omega^2 m}{\rho g A}\right]^2\right\}^{1/2} \tag{5}$$

It is now convenient to introduce a new frequency parameter and also two other relations into equation (5). Substituting for A and m their values for a semi-immersed circular cylinder of length L and radius a and defining the new frequency parameter by

$$\delta = \frac{\omega^2 a}{g} \tag{6}$$

we obtain from equation (5) the normalized force per unit length

$$\frac{F_o}{2\rho g a_o a} = \left\{\left[1 - \frac{\pi}{4}\delta(K + k\right]^2 + \left[(2h)\frac{\pi}{4}\delta\right]^2\right\}^{1/2} \qquad (7)$$

δ is introduced above for convenience in the strip calculation, since both added mass and damping coefficients are given as functions of that frequency parameter in reference [7].

In the strip calculation each spheroid was divided into twenty segments of equal length. In essence the spheroid was represented by twenty cylindrical segments of equal length and various radii. Values of added mass and damping were then assigned to each segment according to the value of δ calculated for each segment. The radius a at the transverse midplane of each segment was used in the calculation of δ for each of several selected values of ω. Using equation (7) the vertical force F_o acting on each segment was calculated. The sum of these forces was then divided by $\rho g a_o A$ (the change in hydrostatic buoyancy force) to yield the normalized force acting on the spheroid. In writing the expression for the change in buoyancy force it has been tacitly assumed that the "wall sided" approximation holds for the small heaving oscillations experienced by the spheroid. The calculations were performed for a sufficient number of values of ω to clearly define the shape of the curves, particularly the location of each minimum.

The force measurements made on the surface spheroids in this experimental study cannot in reality be compared with calculated values, since exact theoretical values for the vertical forces acting on a three-dimensional body heaving in a free surface are known only for the case of a semi-immersed sphere. It can merely be pointed out that the measured forces do not differ greatly from those resulting from the strip calculations. The agreement may reflect the fact that three-dimensional flow aspects are small for the spheroids used here, but the writer is reluctant to voice any strong conclusions because of the unassessed influence of wave reflection from the tank walls. In each case the measured minimum force was less than that calculated. It can be pointed out that the normalized force curves obtained by the strip calculation and plots of the experimental data exhibit much the same behavior as the normalized force curve presented in reference [7] for the two-dimensional circular cylinder. Though several semi-empirical procedures have been developed which correct the strip calculation to some extent for the three-dimensionality of the problem, no attempt was made to apply them to the forces calculated here.

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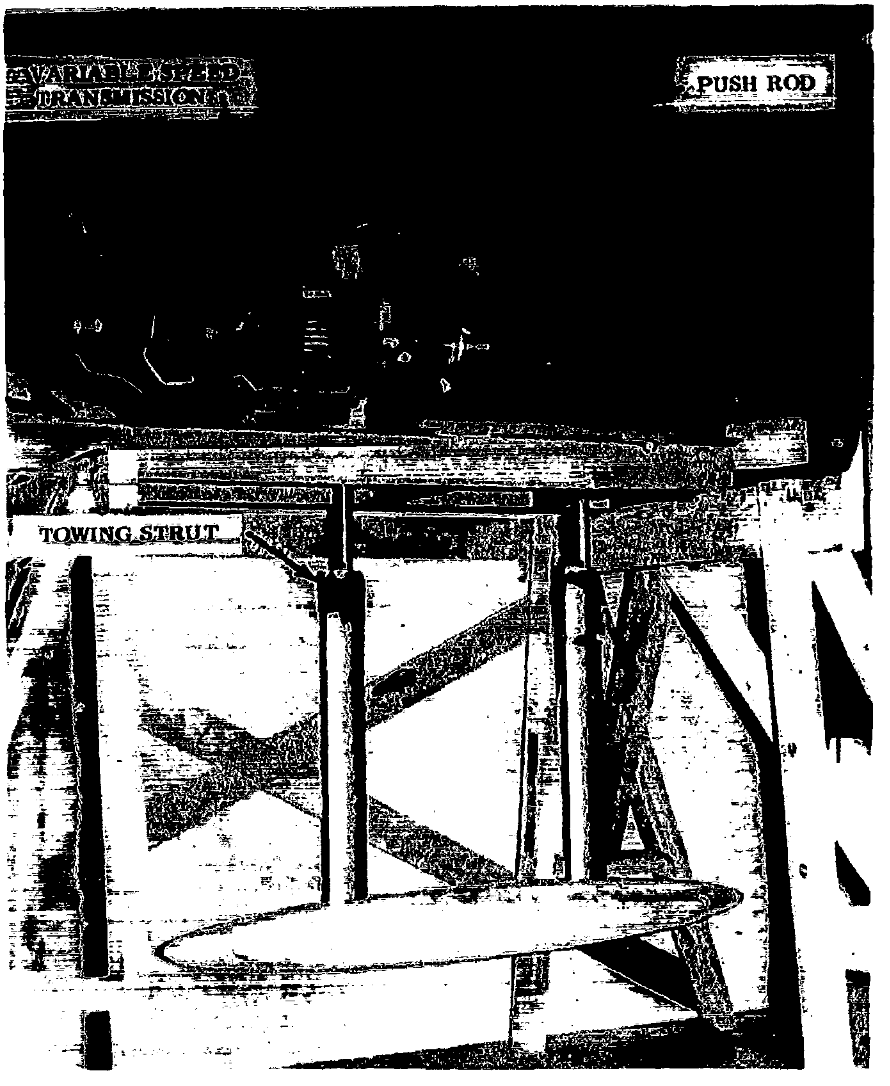
ACKNOWLEDGEMENTS

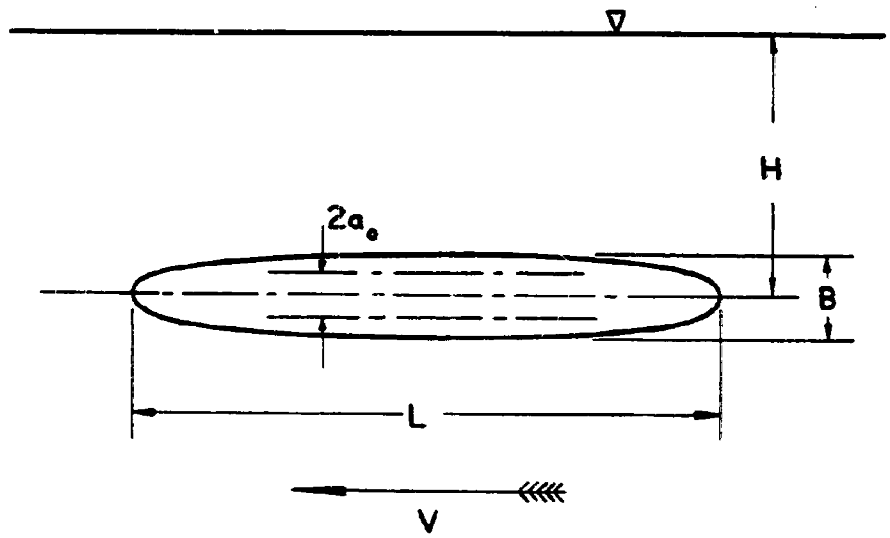
The valuable guidance of Professor J. R. Paulling, Jr., in the design of the motion generator and in the execution of the experimental investigation is gratefully acknowledged. The writer wishes to express his gratitude to two other people: his wife, who carried out the lengthy strip calculations, and typed the manuscript, and Dr. William R. Porter, who offered many helpful suggestions concerning the experiments.

The work was jointly sponsored by the Naval Architecture Department, University of California; a graduate student research grant from the University of California; and, especially, by the United States Navy, who through their Office of Naval Research Graduate Study Program made it possible for the writer to attend the University of California for two years.



Photograph of Planar Motion Generator with Spheroid Installed in Heave Test Position

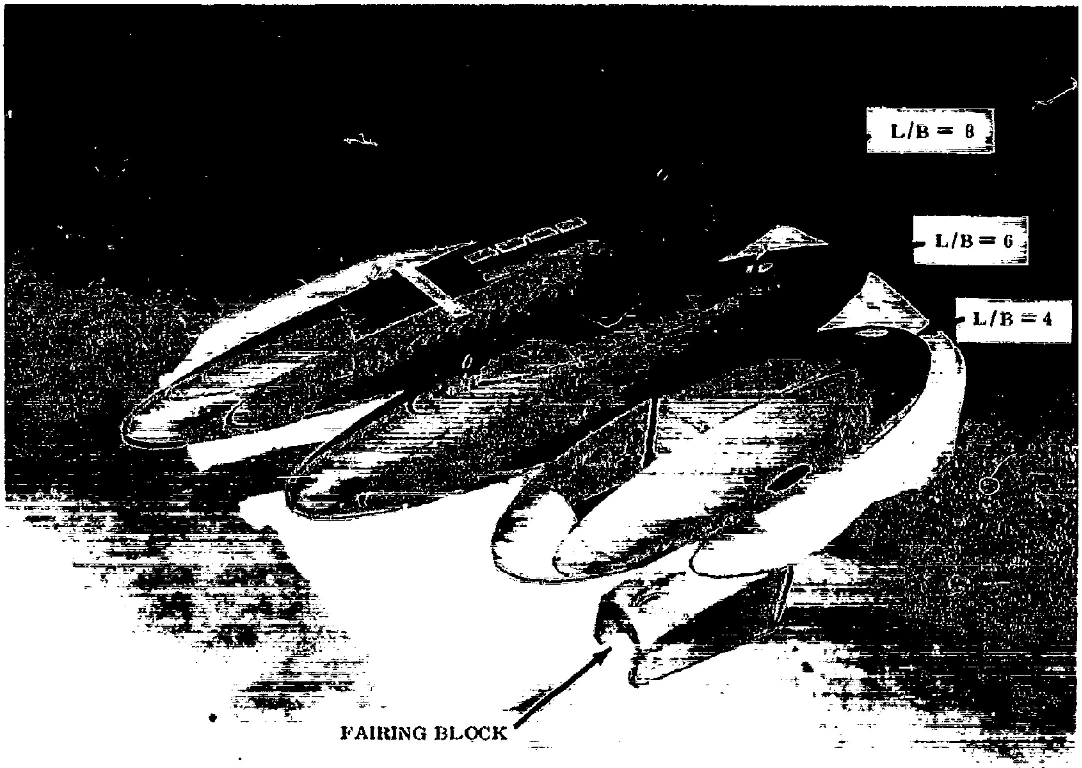
Figure 1



MODEL	L(ins)	B(ins)
A	48	6
B	39.6	6.6
C	30.2	7.55

Figure 2

Geometry of the Experimental Setup and Models



Photograph of Prolate Spheroids used in Experimental Investigation

Figure 3

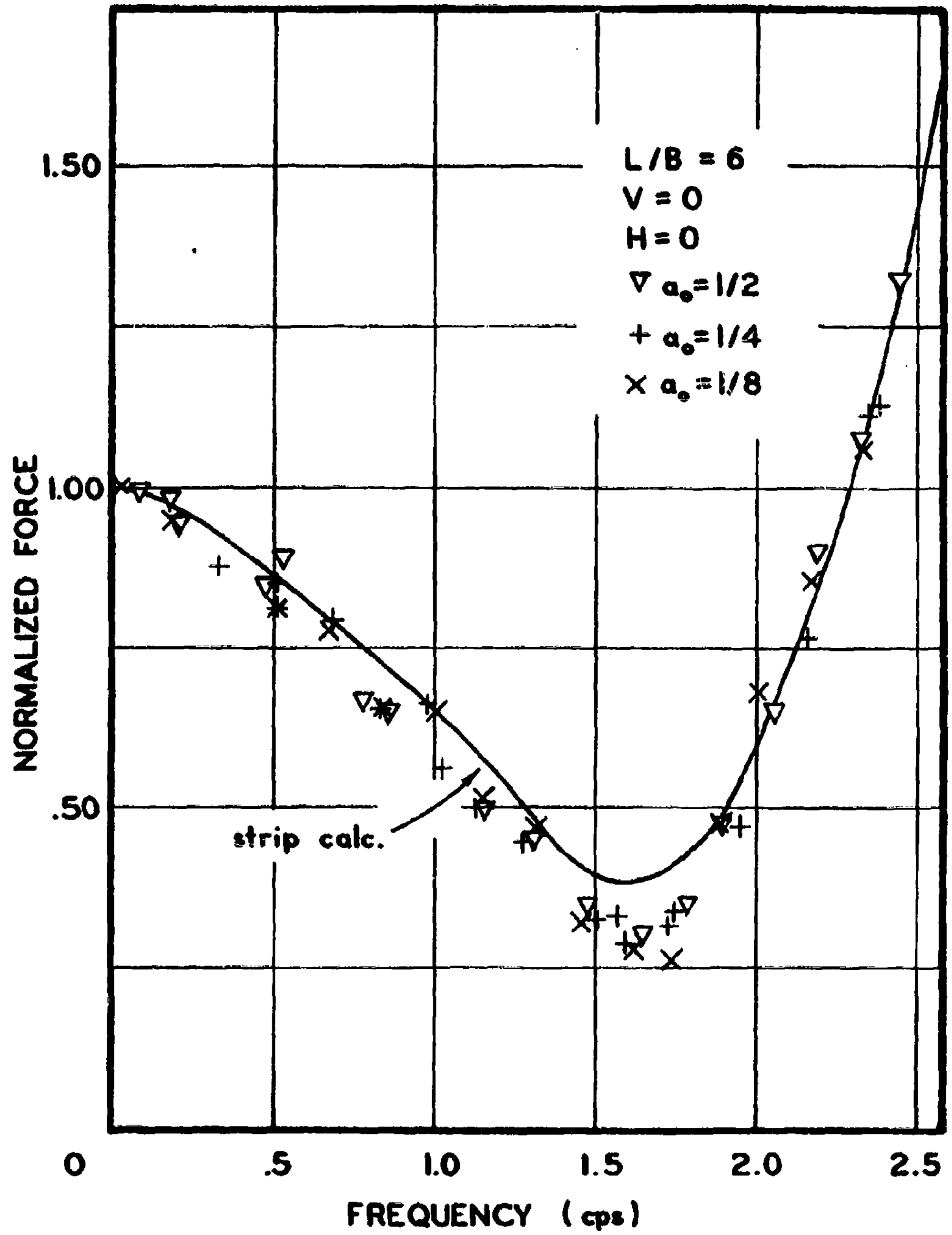


Figure 4

Normalized Heaving Force Acting on Semi-Immersed Spheroid for Various Frequencies of Oscillation

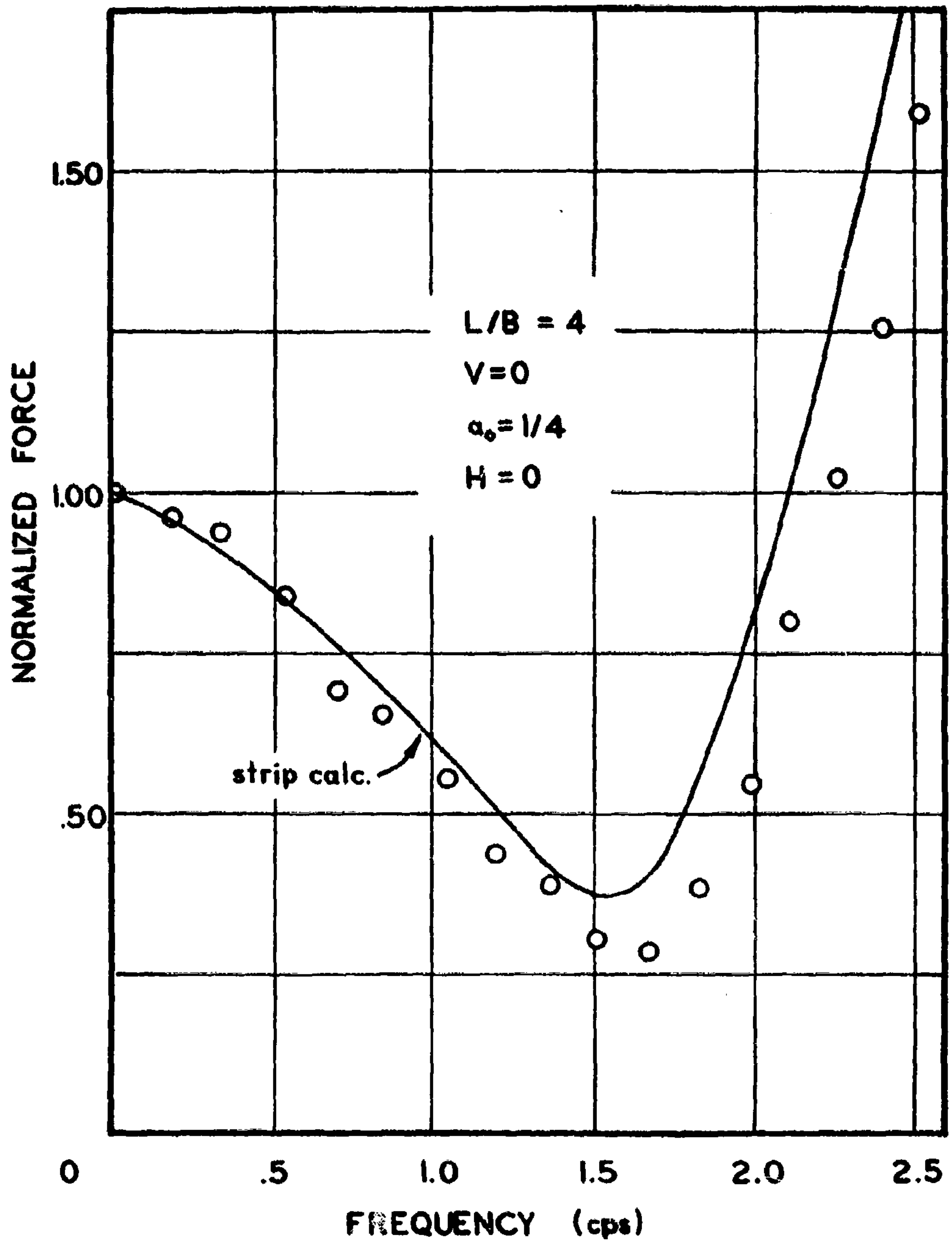


Figure 5.

Normalized Heaving Force Acting on Semi-Immersed Spheroid for Various Frequencies of Oscillation

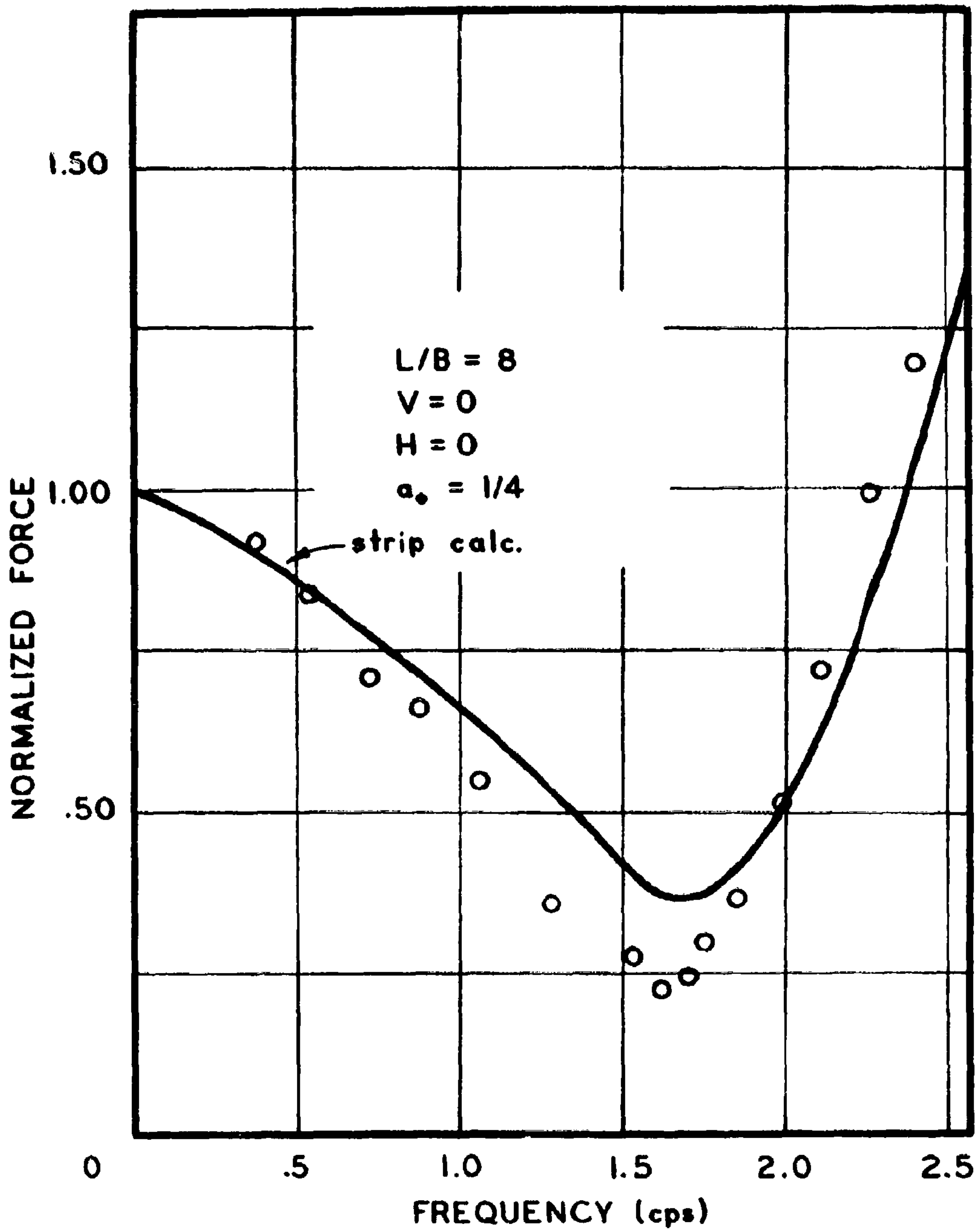
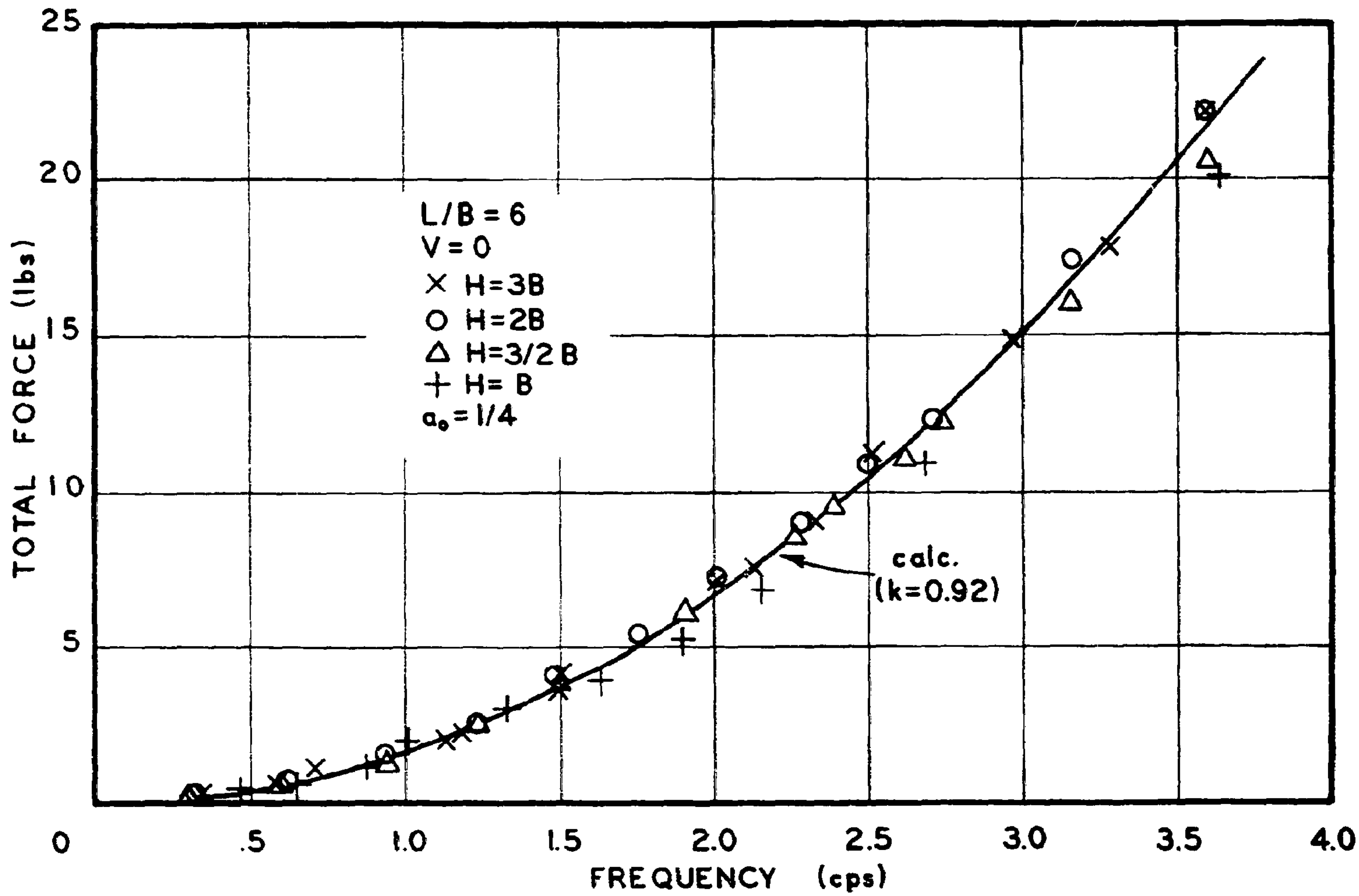


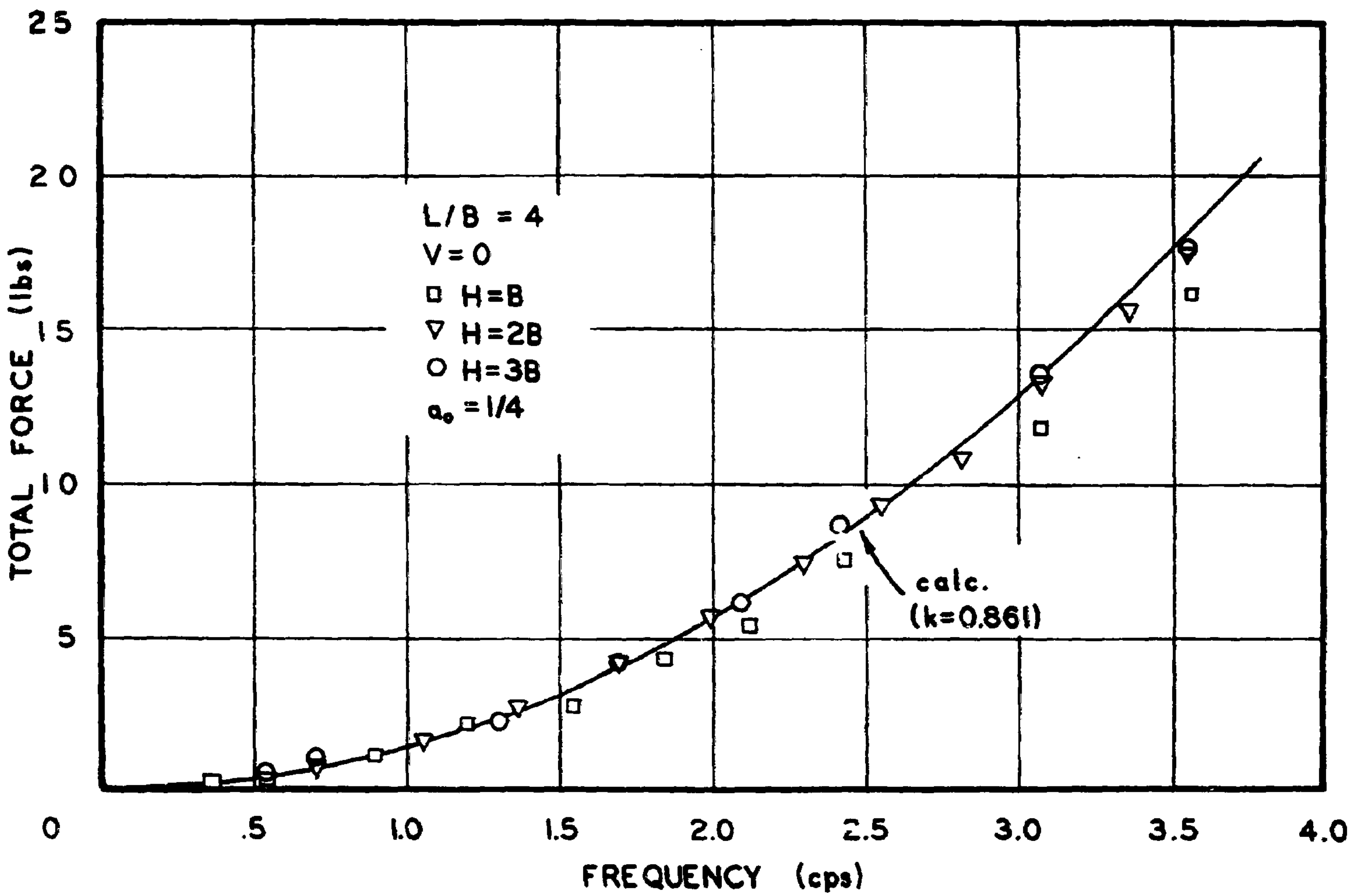
Figure 6

Normalized Heaving Force Acting on Semi-Immersed Spheroid for Various Frequencies of Oscillation



<u>Figure 7</u>

Vertical Heaving Force Acting on Submerged Spheroid for Various Frequencies of Oscillation



<u>Figure 8</u>

Vertical Heaving Force Acting on Submerged Spheroid
for Various Frequencies of Oscillation

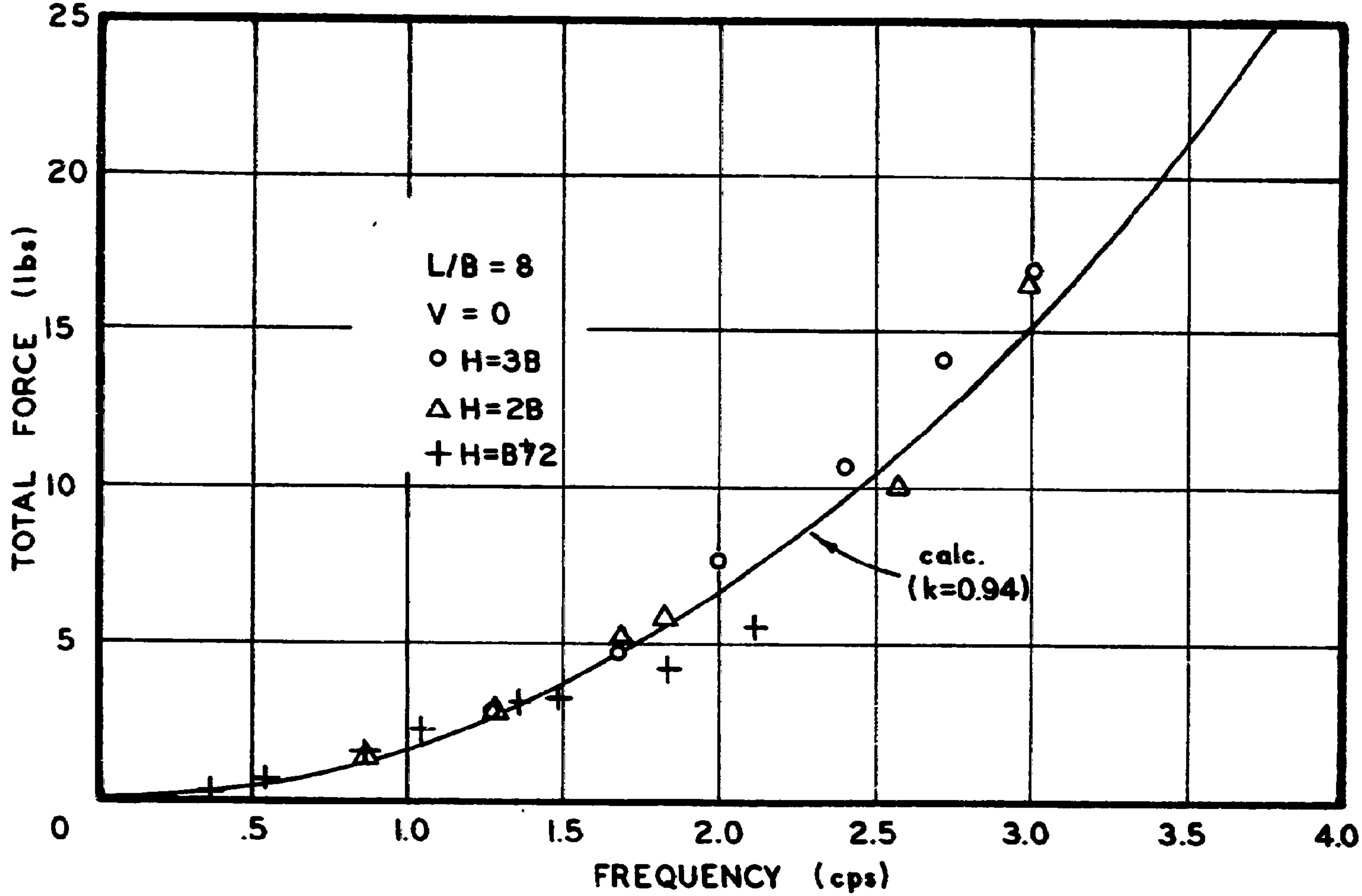


Figure 9

Vertical Heaving Force Acting on Submerged Spheroid for Various Frequencies of Oscillation

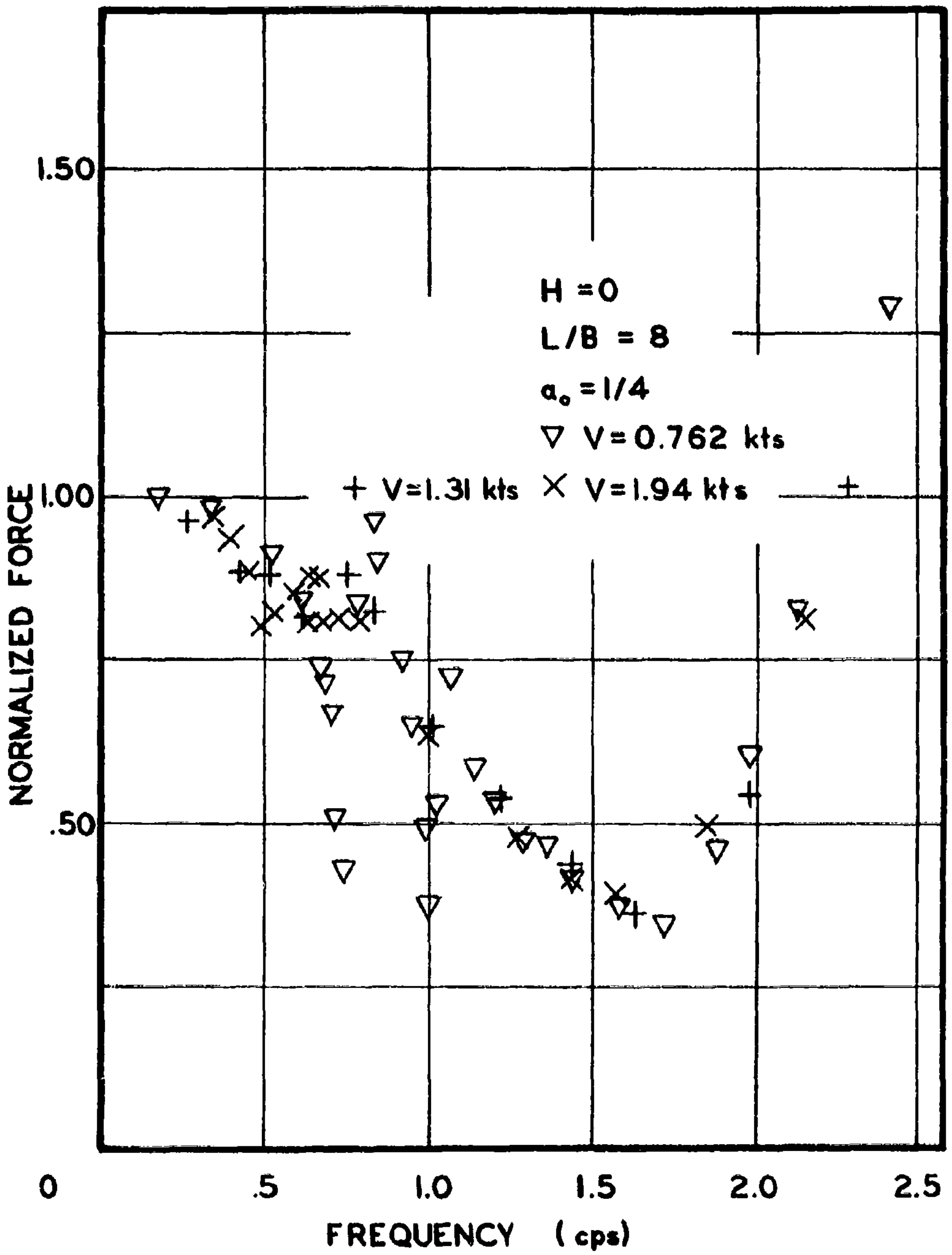


Figure 10

Normalized Heaving Force Acting on Towed Semi-Immersed Spheroid for Various Frequencies of Oscillation

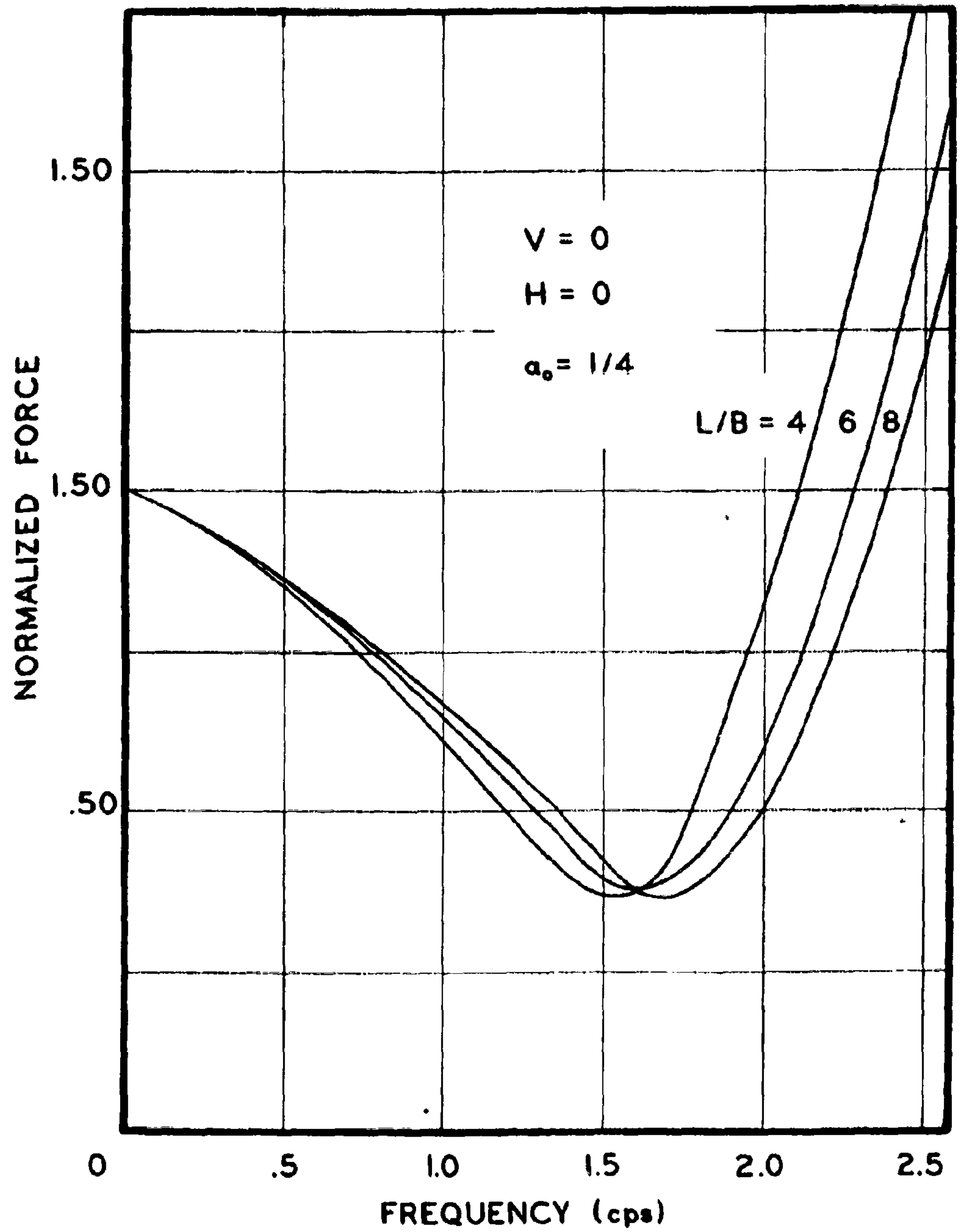


Figure 11

Calculated Normalized Heaving Force Acting on Semi-Immersed Spheroids for Various Frequencies of Oscillation